

Laid-Open Publication**DE 40 04 806 A1**5 **A Method for the operation of a 4-stroke internal combustion engine**

In a method for the operation of a highly charged 4-stroke internal combustion engine, the valve control timings are chosen such that in the indicator diagram
10 the compression phase is shorter than the expansion phase. In the bulk of their operation the inlet valves display the features of a normal inlet closing action, however during the last 30% of valve travel they are closed with a delay, so that the geometrical point of closure is later in time than the bottom dead center.

Description

Technical Field

- 5 The invention relates to a method for the operation of a preferably highly charged 4-stroke internal combustion engine with valve control timings that are chosen such that in the indicator diagram the compression phase is shorter than the expansion phase.

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Prior Art

- A method of this kind is known from the publication by E.F. Obert on "Internal Combustion Engines and Air Pollution" (Intext Educational Publishers, 3rd Edition, 1968). On page 175 of this publication mention is made of the fact that
- 15 it is possible to reduce fuel consumption in internal combustion engines if the expansion phase is longer than the compression phase. A working process of this sort can be achieved, for example, by control of the inlet and/or exhaust valves.

- 20 Further, on page 179 of this work, the idealized working method of an internal combustion engine is portrayed, and on page 215 we read that by increasing the charge, on the one hand higher performance can be achieved and on the other hand the fuel consumption can be reduced by means of positive adjustments to the charge exchange cycle.

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Finally, on pages 616-617 of this publication, we are told that supercharged engines can with advantage have low compression ratios in order to reduce the maximum pressure.

- 30 These observations lead to the conclusion that at full load in an internal combustion engine the expansion phase should be longer than the compression phase and that the effective compression during the compression phase must be low. On the other hand, the starting characteristics of the

internal combustion engine are improved by high compression ratios. Hence the engine should have a variable compression ratio.

In fact a number of different solutions have been put forward hitherto, for
5 example the BICERA Variable Compression Engine (see Obert, page 618), or
the so-called Miller Systems (U.S. Patent No. 2,817,322). The latter are
distinguished from the usual charging systems by the variable closing point of
the inlet valve, by means of which the inlet valve is closed increasingly early
with increasing load, so that the cylinder receives only an incomplete charge of
10 air. However, all the known systems to date have the disadvantage of a
relatively complicated adjustment mechanism.

Disclosure of the Invention

15 It is here that the invention proposes to offer assistance. It is based on the need
to create a method of the kind referred to above that will achieve the stipulated
reduction in fuel consumption without additional mechanical adjustment devices
for the valves and without adversely affecting the starting characteristics
thereby.

20 According to the invention this is achieved in such a way that during the bulk of
their operation the inlet valves display a normal inlet closing action and that
preferably during the last 30% of valve travel the inlet valves are closed with
delay, so that the geometrical closure point is later in time than the bottom dead
25 center.

The particular advantage of the invention can be seen in the simplicity of the
measure, i.e. the desired objective is achieved simply by selecting special valve
lift curves, especially at the end of their inlet closing action. By means of the
30 special control timings the energy surplus can be recovered by gas exchange
processes - especially with two-stage charging, for example by means of
exhaust gas turbochargers.

Brief description of the drawings

The drawings represent an example of an embodiment of the invention in simplified form. The invention is explained by means of the control and indicator diagrams for a fully-charged, moderately fast-running 4-stroke Diesel engine, in which:

- Fig. 1 shows the profile of the valve lift (H_v) represented as a function of the crankshaft position ($\text{Grad KW} = \text{degrees crankshaft}$);
- Fig. 2 shows the cylinder pressure (P_z) as a function of the ratio of cylinder volume to piston-stroke volume (V_z/V_h) during the charge exchange of a Diesel motor;
- Fig. 3 shows the cylinder pressure (P_z) as a function of the ratio of cylinder volume to piston-stroke volume (V_z/V_h) during the inlet stroke of a Diesel engine.

Best Mode for Carrying out the Invention

- Not shown is the already familiar layout of an engine charged via a turbo charger. Let it be assumed that the engine is charged with two chargers, the compressors of the chargers being arranged in series, one behind the other, with or without intercooling.
- The inlet valves of the 4-stroke internal combustion engine follow a closing path corresponding to curve A in Fig. 1, which shows the valve lift as a function of the crankshaft position. Curve A is characterized by the delayed closing of the valve in comparison with a normal curve, corresponding to the prior art, as shown in curve B in Fig. 1. At normal working speed the cylinder charging thus corresponds to an inlet closing action C in Fig. 1, which has a closing point lying before the bottom dead center (BDC /UTP). At engine start-up, by contrast, the

effective cylinder charging corresponds to the closing action B in Fig. 1 and is thus greater than the cylinder charging according to curve C.

Curve A corresponds in its path to curve B, but is displaced and is closed with delay at the end of the closing action. With normal valve lift curves, Curve C in Fig. 1 would show the form of the end of the closing action and hence the valve in Fig. 1 would already be closed at 515° KW. Curve A corresponds to a valve that closes fully only later, for example at the 565° KW shown in Fig. 1.

The invention works as follows: At high engine revs during the last 30 per cent of valve travel of Curve A, only a small amount of air can now flow through the opened valve, since the lift is small and the time available is short. As a result of this procedure the throttling effect on the air flow is great; the cylinder charging achieved as a final result corresponds approximately to the valve closing action according to Curve C in Fig. 1.

When starting the engine, however, the engine revs are low and the above-mentioned throttling effect of the airflow cross-section is slight; hence the cylinder charging corresponds roughly to the maximum achievable volume, as dictated by the piston stroke.

Thus the stipulated objectives are attained, namely high compression ratio when starting the engine and lower compression ratio during normal running, the expansion phase in the latter case being longer than the compression phase.

On the basis of simulation calculations for a Diesel engine running at moderate speed, the effect of this valve control can be explained by reference to Fig. 2.

This shows the cylinder pressure (P_z) as a function of the ratio of cylinder volume to piston-stroke volume (V_z/V_h) during the charge exchange of the Diesel engine. Curve A once again corresponds to operation with the inlet valve closing action in accordance with Curve A in Fig. 1. Curve B corresponds to

operation with inlet valve closing action in accordance with Curve B in Fig. 1. What is noticeable is the expansion at the end of the inlet process of Curve A (indicated with * in Fig. 2). In this way the cylinder charge is reduced. In order to obtain the maximum cylinder pressure during combustion the charge pressure (indicated by P_s in Fig. 2) - and thereby also the charge exchange work - is increased, assuming that the charging efficiency is sufficiently large. The charge pressure P_s during operation - which corresponds to Curve B - is considerably lower, but the cylinder pressure at the end of the charge exchange is equal to that of Curve A. This again indicates that the cylinder charge during the operation is reduced to Curve A.

Thus the special valve control results in a reduction of the fuel consumption because on the one hand the compression phase is shorter than the expansion phase and on the other the charge exchange work is increased, as a comparison of curves A and B in Fig. 2 shows.

On the basis of simulation calculations for the same engine, the starting characteristics of the new valve control methods can be explained with reference to Fig. 3. This shows the cylinder pressure (P_z) as a function of the ratio of cylinder volume to piston-stroke volume (V_z/V_h) during the induction stroke of the Diesel engine. Curve A corresponds to operation with the inlet valve closing profile in accordance with Curve A in Fig. 1; Curve B corresponds to operation with the inlet valve closing profile as in Curve B in Fig. 1. The charge pressure is the same both in Curve A and in Curve B. What is notable is that the cylinder pressure at the end of the charge exchange is the same for Curve A as for Curve B. This indicates that the cylinder charge is the same when operating according to Curve A and Curve B and that when starting up the engine a high compression ratio is achieved.

Patent Claim

A Method for the operation of a preferably highly charged 4-stroke internal combustion engine with valve timings that are chosen such that in the indicator
 5 diagram the compression phase is shorter than the expansion phase,
 characterized in that in the bulk of their operation the inlet valves display the
 features of a normal inlet closing action and that preferably during the last 30%
 of valve travel they are closed with a delay, so that the geometrical point of
 closure is later in time than the bottom dead center.

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3 pages of drawings attached

15 Translation of abbreviations (legends in drawings)

Fig 1:

Hv	=	valve lift
Grad KW	=	degrees crankshaft
20 UTP	=	BDC

Figs. 2& 3:

Pz	=	cylinder pressure
Vz	=	cylinder volume
25 Vh	=	piston-stroke volume
Ps	=	charge pressure